IN THE UNITED STATES PATENT & TRADEMARK OFFICE

SPECIFICATIONS AND CLAIMS OF PATENT APPLICATION

Power Cogeneration System And Apparatus Means For Improved High Thermal Efficiencies and Ultra-Low Emissions

BACKGROUND OF THE INVENTION

When Brayton Simple Cycle gas turbines operate as mechanical power drive sources to electric generators and other mechanically driven devices, atmospheric air is compressed and mixed with hydrocarbon gases or atomized hydrocarbon liquids for the resulting mixture's ignition and combustion at constant pressure. To produce power, the hot combustion and working motive fluid gases are expanded to near atmospheric pressure across one or more power extraction turbine wheels, positioned in series.

The majority of Brayton simple open-cycle aero-derivative-style Low-NO.sub.x art gas turbines are predominantly presently limited in achieving shaft output horsepower rating with 26% to 39% thermal efficiencies, whereas most simple cycle industrial-style Low-NO.sub.x art gas turbines are predominantly presently limited in achieving shaft output horsepower rating with 27% to 34% thermal efficiencies. The industrial turbine higher efficiencies are achieved when the gas turbines operate with

compressor ratios ranging from 10 to 15 and predominant first stage turbine inlet temperatures ranging from 2100° to 2300° F.

Existing gas turbines employ combustion chamber air/fuel combustion chemical reactions, wherein the elements of time and high peak flame temperatures increase the presence of disassociation chemical reactions that produce the fugitive emissions of carbon monoxide (CO) and other chemical reactions that produce nitrogen oxides (NO.sub.x).

The best available applied turbine low NO.sub.x combustion technology for limiting gas turbine NO.sub.x emissions, using stiochiometric air/fuel primary combustion reaction chemistry means, still results in the production of NO.sub.x and CO that are no longer acceptable for new power or energy conversion facilities in numerous states and metropolitan environmental compliance jurisdictions. With the conventional gas turbine's use of compressed atmospheric air as a source of oxygen (O.sub.2) which acts as a fuel combustion oxidizing reactant, the air's nitrogen (N.sub.2) content is the approximate 78% predominant mass component within the cycle's working motive fluid. Due to its diatomic molecular structure, the nitrogen molecules are capable of absorbing combustion heat only through convective heat transfer means resulting from their collisions with higher temperature gas molecules or higher temperature interior walls of the combustion chamber.

Despite the very brief time it takes for conventional gas turbine to reach a average molecular primary flame combustion zone gas equilibrium temperature of less than 2600° F within its combustion chamber, there are sufficient portions of the combustion zone gases that experience temperatures in excess of 2600° F to 2900° F for an ample period of time for the highly predominate nitrogen gas to enter into

chemical reactions that produce nitrogen oxides. The same combined elements of time and sufficiently excessive high flame temperature permit carbon dioxide to enter into dissociation chemical reactions that produce carbon monoxide gas.

To achieve a goal of greatly reducing a turbine power unit's NO.Sub.x and CO fugitive emissions, it is necessary to alter both the fuel combustion chemical reaction formula and the means by which acceptable combustion flame temperatures can be closely controlled and maintained within a power turbine unit's fuel combustion assembly. Maintenance of an acceptably low selected fuel combustion peak gas temperature at all times and throughout all portions of within the combustion assembly, requires a change in the means by which the heat of combustion can be better controlled and more rapidly distributed uniformly throughout the gases contained within the fuel combustion assembly.

To achieve a goal of significantly reducing a turbine power cogeneration system's mass emission rate of the "greenhouse gas" (carbon dioxide) by a given percentage, it is necessary to proportionally increase the thermal efficiency of a power cogeneration system which therein proportionally reduces the amount of combusted hydrocarbon fuel required to provide the energy conversion into a given amount of required work and usefully applied residual heat energy.

It has been well known and practiced for decades that higher humidity air and injected water or steam commingled with conventional air combustion gases increases combustion flame speeds and fuel combustion thermal efficiencies within gas turbines and other fuel combustion burner apparatus using air/fuel combustion. It has also been well known and practiced that partially re-circulating combustion flue gases containing carbon dioxide back into a combustion chamber results in a reduced level of nitrogen

oxides within the fuel combustion exhaust gases. Due to the high temperatur s and speed of completed fuel combustion, the scientific community has been unable to reach a consensus as to precisely what series of altered chemical reactions occur when water vapor and/or carbon dioxide is introduced into a turbine combustion chamber.

Oxy-fuel combustion burners have been employed for many years in the steel and glass making industries to furnish desired 3000+ degree Fahrenheit combustion gas temperatures into furnaces to avoid the production of high NO.Sub.x emissions (but at the expense of high CO emissions). Both the present air separation art methods' high energy costs of producing acceptable combustion grade oxygen, and the lack of devised combustion system methods to control preset desired oxy-fuel combustion burner or combustion chamber uniform maximum temperatures, have curtailed oxy-fuel combustion applications within present energy conversion facilities.

Conventional gas turbines must be de-rated from their standard ISO horsepower or kW ratings at ambient temperatures exceeding 59° F, or at operating site altitudes above sea level. Thus, during summer's peak power demand periods, when the ambient temperature can increase to 95° F, 20% to 25% horsepower derations of a conventional gas turbine's ISO rating can occur. It is obviously desirable that a power turbine/generator unit within a cogeneration system not be susceptible to such on-site ambient temperature derations when peak power demands occur.

The current and future projected increasing costs of purchased utility electric power and natural gas (or liquid hydrocarbon fuel) and the accepted projected future trend in the future of "distributed power" facilities, coupled with present and future environmental constraints on fuel combustion exhaust emissions, will make it commercially mandatory that such "distributed power" facilities have the combined

attributes (at the minimum) of combined ultra-low NO.sub.x and CO exhaust emissions and substantially higher thermal efficiencies than offered by current art power cogeneration systems. It can be expected that the number of new turbine powered 'cogeneration system' facilities in the world will be significantly greater than the number of turbine powered 'combined-cycle' facilities that are devoted purely to the production of electric power. The referenced 'cogeneration system facilities' are not new in concept. Such facilities became highly popular in the 1970's (then referred to as 'Total Energy Plants') and were aggressively promoted by many natural gas utilities. Reciprocating gas engine-driven generator sets were the predominant producers of prime power and utilized waste heat. These 'Total Energy Plant' facilities efficiently provided electricity, hot water or steam for domestic hot water and building heating requirements, and chilled water for air conditioning. 'Total Energy Plants' were widely applied to serve hospitals, universities, large office buildings or building complexes, shopping centers, hotels, food processing plants, multi-shift manufacturing and industrial facilities, etc. The 50 plus years old predecessor to the 'Total Energy Plant' concept was the central electric power and steam plants that continue to currently serve some large eastern US cities, and more predominantly European cities and metropolitan areas. Predominantly, 'Total Energy Plants' and current cogeneration facilities have had less than 100 psig utility supplies of natural gas available to their facilities.

Summary of the Invention

To achieve both power turbine ultra-low NO.sub.x and CO exhaust emissions (as well as reduced "greenhouse gas" (CO.sub.2) and enhanced simple-cycle operating

thermal efficiencies, the inventor's AES gas turbine power cycle system and apparatus is described in U.S. Patent #6,532,745 dated March 18, 2003. The cited invention's further described partially-open gas turbine cycle contains multiple heat recovery devices for transferring waste heat to varied process gases and steam resulting in a cogeneration facility overall maximum thermal efficiency that "may approach 100%".

The present invention describes the means by which the cited partially-open AES turbine power cycle system and apparatus can be incorporated into a simplified and improved gas turbine cogeneration system having simplified apparatus means and that can further achieve increased turbine cogeneration system thermal efficiencies which exceeds 115%.

The present invention further describes the alternate system and apparatus means for the cited improved partially-open turbine cogeneration system that can be employed within a desired power cogeneration system design, the said alternate system and apparatus means incorporating portions of the AES heater cycle system and apparatus content cited in the inventor's U.S. Patent application 10/394847 filed March 22, 2003 and titled "Partially-Open Fired Heater Cycle Providing High Thermal Efficiencies and Ultra-Low Emissions". The addition of this alternative to the presented turbine based cogeneration system, as later further described and shown in Figure 2, can increase the presented cogeneration system's overall thermal efficiency to greater than 115%.

The commercial viability of achieving maximum reductions in the presented invention's enhanced cogeneration system's fuel operating costs (with accompanying reduced NO.sub.x, CO, and CO.sub.2 exhaust emissions is assured by the presented invention's oxy-fuel combustion system's access to a facility-provided ultra-high electric

energy efficient modular air separation system, such as presented in the inventor's U.S. Patent Application 10/658157 dated September 9, 2003 and titled "Pure Vacuum Swing Adsorption System and Apparatus".

To achieve the cogeneration system's ultra-low fugitive exhaust emissions, the cited partially-open power cogeneration system and apparatus means provides a continuous controllable mass flow rate of recycled superheated vapor-state predominant mixture of carbon dioxide (CO.sub.2) and water vapor (H.sub.2 O), in identical mixture Mol percent proportions as each occurs as products of chemical combustion reactions from the gaseous or liquid hydrocarbon fuel employed.

To achieve the cogeneration system's ability to employ gaseous hydrocarbon fuels, other than gas utility distribution quality natural gas, gaseous fuels (containing toxic and/or difficult to combust hydrocarbon molecular gases) can be rapidly carried through useful heat conversion and/or completed incineration with the inventions provided system and apparatus means to control the primary and secondary combustion zones temperature. Whereas the invention example system's presented recycle exhaust gas flow rates and temperatures are capable of producing 1800°F combustion temperatures to the turbine assembly (while maintaining herein described high thermal efficiencies and ultra-low emissions), the preferred example 2400° F primary and outer secondary zone combustion temperature provides a desired 7.585 greater chemical reaction rate than that occurring at 1800° F. As repeatedly verified by John Zink Research in applied research, the reaction rate formula is:

$$\text{Reaction Rate Increase} = (N) = \frac{[(2400^\circ\text{ F}+460) \div (1800^\circ\text{ F}+460)] - 1}{.035}$$

Provided herein is both a partially-open turbine power cogeneration system with apparatus means for use therein of either modified conventional gas turbine unit configurations, or alternative AES turbine assembly unit apparatus configurations that can utilize separate existing low cost mechanical equipment components and burner assemblies which are predominantly not designed for, nor applied to, the manufacture of conventional gas turbines nor the said components and burner assemblies incorporation into facility designs of current technology gas turbine cogeneration systems (or combined-cycle systems).

The invention's combined employed cited partially-open gas turbine cycle system and apparatus and alternative added cited AES heater cycle system and apparatus portion into the present invention therein provides for a commonly 'shared non-air' working motive fluid means that is essential to the reduction of NO.sub.x, CO, and CO.sub.2 mass flow fugitive emissions from those of conventional Low-NO.sub.2 designed gas turbines and other conventional fuel combustion burner devices that can be applied within existing art power cogeneration systems.

It is an objective of the present invention's improved power cogeneration system and apparatus means to provide a new benchmark standard for Best Available Technology (BAT) in achieving combined highest thermal efficiencies, lowest emissions, and lowest auxiliary facility operating power consumptions within a overall power cogeneration facility.

It is a further objective of this invention to provide the means by which the power cogeneration system's production of steam or hot water, and/or the heating of process fluids, is not limited by the amount of a turbine/generator or mechanical drive train's

available waste h at derived from a given production level of electric power or mechanical horsepower.

It is a further objective of this invention to provide the means by which the power cogeneration system's presented alternate apparatus can comprise unconventional power train unit components that can be adapted to individual unit power generator ratings of 200 kW to 30 MW+ to satisfy most cogeneration facility installed individual unit rating requirements.

It is a further objective of this invention to provide the collective means by which deviations from the presented invention's example operating conditions can be made to best accommodate a facility designer's incorporation of existing models of other facility auxiliary equipment that can be further incorporated into a specific design of cogeneration facility, such as currently manufactured adsorption chillers or mechanically-driven refrigeration chillers that have been conventionally or similarly applied in related waste heat recovery power facilities for over 30 years.

It is a further objective of the present invention's cogeneration system and apparatus means to accomplish both a highly accelerated fuel combustion process and the added means to control a preset maximum primary combustion zone temperature generated from its applied oxy-fuel combustion process. This satisfied objective eliminates the elements of time and high degree of temperature that is required for endothermic dissociation chemical reactions to occur that produces both NO.sub.x and CO within the combustion product gases.

It is a further objective of the present invention of improved system and apparatus means that an AFE power system modified conventional gas turbine assembly or unconventional re-configured turbine train assembly can be capable of achieving 20 to

25% or greater thermal efficiencies than conventional art simple-cycl gas turbines operating alone, ~~or~~ and 40 to 50% greater thermal efficiencies than conventional gas turbine power cogeneration facilities.

It is a further objective of the present invention of improved system and apparatus means that the cited incorporated partial-open gas turbine cycle system and apparatus means of preferred high efficiencies can employ (but not limited to) gas compression ratios of 2.4 to 6.4 (2.1 to 6.5 Bar operating pressure) as compared to conventional gas turbines having compression ratios of approximately 9 to 35.

It is a further objective of the present invention of improved system and apparatus means that the cited partial-open gas turbine cycle system and apparatus can provide the maximum cogeneration thermal efficiencies with facility fuel gas supply pressures of less 100 psig (6.9 bar).

It is a further objective of this invention to provide the means wherein, during a steady-state power operation, that the atmospheric vented and open cycle portion of the cogeneration system exhaust mass flow can be approximately 5 to 8% of the total working fluid mass flow rate as contained within the closed portion of its turbine power cogeneration system.

It is a further objective of this invention to provide the means whereby both the cited partial-open AES gas turbine cycle system and apparatus as applied within the present invention of improved cogeneration system efficiency, and the alternative cogeneration system apparatus means described herein, can include appropriate safety sensor and system fluid flow control device means. Both the presented invention's cogeneration system and apparatus component means and the separately associated cogeneration power plant auxiliaries can be monitored and controlled for safe operation, as well as

having provided means for controlling the cogeneration system's closed system fluid flows in response to changes in electric power generation demands and effective heat extraction demands by supplied steams of steam or hot water, or process fluids.

It is a further objective of this invention to provide the apparatus and control means by which a non-distribution quality of gaseous hydrocarbon fuel (containing toxic and/or difficult to combust hydrocarbon molecular gases) can be rapidly carried through oxy-fuel combustion to a useful heat conversion and/or completed incineration.

First Embodiment

The working motive fluid of this invention's turbine cogeneration system comprises a continuous superheated vapor mixture of predominant carbon dioxide (CO.sub.2) and water vapor (H.sub.2 O) in identical Mol percent ratio proportions as the molecular combustion product components are produced from the combustion of the gaseous or liquid hydrocarbon employed fuel.

Within the predominately-closed portion of the presented cogeneration system and apparatus, the re-circulated turbine exhaust gas is routed from an exhaust gas distribution manifold (the exhaust gas having a small degree of superheat temperature and positive gage pressure supply) into the inlet of the primary recycle compressor. The exhaust gas recycle compression function can be performed by a more typical axial compressor section used for air compression within a conventional gas turbine unit, or it may be a separately means-driven compressor of the axial, centrifugal, or rotating positive displacement type. Either means of compression can incorporate means of flow control available within the compressor or by its driver's varied speed, with flow

changes being initiated by a master system control panel containing programmable microprocessors.

The compressor can increase the recycled turbine exhaust's absolute pressure by a ratio range of only 2.4 to 6.4 to achieve a preferred high simple-cycle thermal efficiency, but the cycle is not limited to operations within these said ratios.

As shown in Table 1, between gas turbine fuel combustion pressures of 45 psia and 75 psia, the AES Cycle thermal efficiencies can range between 35.16% and 43.24%. Between 75 psia and 90 psia oxy-fuel combustion burner assembly pressures (with the common primary recycle compressor and power turbine efficiencies of 84% and stage 1 turbine inlet temperature of 1800° F), the AES turbine cycle system (simple-cycle) efficiencies begins to decline.

TABLE 1

Combustion Operating Pressure	Gas Turbine Gas Inlet Temperature	Gas Turbine Exhaust Temperature	Gas Turbine Net Output Horsepower	Gas Turbine Fuel Rate Btu/HP-Hr.	Thermal Efficiency %*
45 psia	1800° F	1471° F	2859	7237	35.16
60 psia	1800° F	1391° F	3458	5983	42.54
75 psia	1800° F	1331° F	3515	5885	43.24
90 psia	1800° F	1284° F	3406	6075	41.89

*With a 1 Mol/minute methane gas fuel rate

The re-cycled and re-pressurized turbine exhaust gas (hereafter referred to as "primary recycle gas") is discharged from the primary recycle compressor at an increased temperature and pressure through a conduit manifold containing both a side-branch connection and first and second parallel conduit end-branches flow-controlled streams. The conduit manifold side-branch supplied controlled low mass flow stream of

primary recycle gas can be reduced in temperature within an air-cooled exchanger prior to the stream flow's entry into one or more preferred partial pre-mix subassembly contained within each oxy-fuel combustion burner assembly. Within each referred partial pre-mix assembly, the primary recycle gas stream can be homogenously pre-mix blended with the supply stream of predominant oxygen that is also is also supplied to the preferred partial pre-mix subassembly and/or the supply stream of fuel.

The fore-said first and second parallel conduit end-branches flow-controlled streams having end-connectivity respectively to the inlets of first and second headers of the power turbine exhaust gas waste heat recovery unit (WHRU) exchanger of counter-current flow gas to gas heat exchange design. A predominate flow-controlled portion of the power turbine's developed high temperature exhaust is flow-directed through this WHRU exchanger for its heat transfer into the primary recycle gas stream that thereafter is downstream re-admitted into the oxy-fuel fired combustion burner assembly.

This power turbine exhaust gas WHRU exchanger can be capable, with the particular example of a methane fuel combustion chamber pressure of 60 psi absolute and 1800° F first stage power turbine inlet temperature, of raising the temperature of the primary re-pressurized recycle gas within the turbine exhaust gas WHRU exchanger to a maximum 1350° F. With these operating conditions and assumed compressor and hot gas expansion turbine efficiencies of 84%, the desired simple-cycle turbine thermal efficiency of 42.5% can be achieved.

Thereafter, the 1350° F highly superheated and re-pressurized primary recycle gas individual streams are referred to as "working motive fluid". The first controlled stream of working motive fluid can be routed and separately flow-divided as required to the

internal tertiary blending zone contained within each of one or more oxy-fuel fired combustion burner assembly that can be positioned radially about the centerline axis of the power turbine unit assembly. The second controlled stream can be separately flow-divided as required for passage into one or more preferred partial premix sub-assemblies contained within one or more oxy-fuel fired combustion burner assembly. Within the presented power cogeneration system, a lesser flow controlled portion of the total power turbine exhaust flows through the waste heat recovery steam generator (WHRSG) exchanger or waste heat recovery process fluid (WHRPF) exchanger.

Second Embodiment

From the First Embodiment's "the re-circulated turbine exhaust gas is routed from a exhaust gas distribution manifold (the turbine exhaust gas having a small degree of superheat temperature and positive gage pressure supply) into the inlet of the primary recycle compressor", the said re-circulated turbine gas within the exhaust distribution manifold comprises the discharge exhaust gas from a second WHRSG or WHRPF exchanger upstream connected to a re-circulated exhaust gas manifold that conveys the combined turbine reduced temperature exhaust gases from both the WHRU exchanger and the first parallel-positioned WHRSG or WHRPF exchanger into which the total gas turbine high temperature exhaust is first inlet-connected.

Either the second WHRSG or second WHRPF exchanger can perform the initial heating of supplied streams from either a facility's steam or hot water feed circuit or a process fluid stream prior to either of these streams being further downstream flow-connected to the fore-described high temperature turbine exhaust gases first WHRSG exchanger or WHRPF exchanger.

Third Embodiment

From the First Embodiment cited re-circulated turbine exhaust from the exhaust gas distribution manifold supplied to the inlet of the primary recycle compressor, the exhaust gas distribution manifold has a end manifold alternative system connection point and two side-branch flow delivery connections. The first side-branch conduit provides the greatly predominant flow of re-circulated into the inlet of the recycle compressor, and the second side-branch conduit directs the controlled flow of excess of re-circulated turbine exhaust gases to atmosphere during steady-state operation of the presented system. This flow of excess re-circulated turbine exhaust gases to atmosphere constitutes the "Open Portion" of the presented partial-open power cogeneration system. The system steady-state condition's controlled mass flow rate in which the re-circulated turbine exhaust is vented to atmosphere is equivalent to the combined mass rates at which the fuel and the predominant oxygen gas streams enter the invention's provided oxy-fuel combustion system and apparatus means.

Fourth Embodiment

From the First Embodiment cited "The second controlled stream can be separately flow-divided as required for passage into one or more preferred partial pre-mix sub-assemblies contained within one or more oxy-fuel combustion burner assembly", each partial pre-mix sub-assembly has the following introduced controlled streams: fuel; a predominant oxygen stream which originates from an adjacent facility area containing a preferred highly electric energy efficient modular air separation system; First Embodiment described air-cooled primary recycle gas; and second stream of working motive fluid. These individual flow controlled conduit streams at differential pressures and velocities are collectively admitted through their respective pre-mixer

inlet conduit means for preferred selective pre-mixing and homogeneous blending at points of admittance into the primary combustion flame zone and outer secondary zone within each oxy-fuel combustion burner assembly.

To establish primary combustion temperatures that do not exceed the example preferred maximum 2400 F, one of several possible acceptable designs of pre-mix sub-assembly is one of wherein the oxy-fuel combustion burner assembly can incorporate both a primary oxy-fuel combustion flame zone and a secondary outer zone wherein a predominant portion of the fore-described second stream of working motive fluid is introduced into a outermost flow annulus area surrounding the homogeneous mixture admitted from each pre-mix sub-assembly into the said primary combustion flame zone for ignition. The secondary outer zone introduced working motive fluid can thereby provide a closely positioned rapid heat-absorbing mass shrouding means around each primary combustion flame zone developed within the oxy-fuel burner assembly. This flame shrouding means can enable the radiant heat energy emanating from the binary gas molecules within the combustion flame to be rapidly distributed to and absorbed uniformly by the described shroud's contained identical binary gaseous molecules at the speed of light rate of 186,000 miles per second. The resulting equilibrium temperature within each oxy-fuel burner assembly's primary combustion flame zone and secondary zone, based on the controlled flow rate of the second stream of working motive fluid into the oxy-fuel combustion burner assembly, can be established as being equal to a preset desired example of a maximum 2400° F or other desired preset temperature that is substantially less than the temperature at which NO.Sub.x and CO can be formed during endothermic disassociation chemical reactions. The example maximum 2400 F merely represents a conservative maximum temperature to totally

avoid the slightest potential of any combined production of extremely small trace amounts of NO.sub.x and companion larger amounts of CO.

Fifth Embodiment

From the First Embodiment cited "The first controlled stream of working motive fluid can be routed and separately flow-divided as required to the internal tertiary blending zone contained within each of one or more oxy-fuel combustion burner assembly that can be positioned radially about the centerline axis of the power turbine assembly", the first flow stream can be routed and separately flow-divided as required to the internal tertiary blending zone contained within each of one or more oxy-fuel fired combustion burner assembly that can be positioned radially about the centerline axis of the power turbine assembly. The tertiary blending zone flow of working motive fluid can be introduced into an oxy-fuel combustion burner assembly's inner annulus area between the burner assembly's outer casing and an inner liner surrounding each primary oxy-fuel combustion flame zone and outer secondary zone, followed by its flow emanation into the burner assembly's tertiary blending zone chamber area through openings in the liner. This tertiary zone introduced mass flow of superheated working motive fluid blends with the example maximum 2400° F equilibrium temperature combined gases emanating from the burner assembly's primary oxy-fuel combustion flame zone and its outer secondary zone to thereby produce a resultant example 1800 F final oxy-fuel burner assembly exhaust equilibrium temperature to the power turbine assembly. The equilibrium temperature of the final oxy-fuel burner assembly exhaust gases is not limited to 1800°F, and may be controlled by the introduced tertiary mass flow rate and/or fuel mass flow rate to establish any other higher or lower selected operating

temperature. The example 1800°F temperature is chosen to coincide with 10 year old proven power turbine blade metallurgy technology for continuous operation.

Within the one or more hot gas expansion turbine stages, the oxy-fuel combustion burner assembly's pressurized and highly superheated gases is expanded to create useful work in the conventional form of both turbine output shaft horsepower and (in the case of a conventional modified gas turbine unit configuration) internal horsepower to additionally direct-drive the primary recycle compressor. In a conventional 2-shaft style of gas turbine, the primary recycle compressor is shaft-connected to the high-pressure stage section of the power turbine assembly, and the low pressure section of the power turbine assembly provides the turbine power output power to driven equipment. The expanded exhaust gases exit the power turbine assembly at a low positive gage pressure and are further conveyed through conduit means to the fore-described WHRU exchanger and adjacent parallel-position WHRSG or WHRPF exchanger as further described later and shown in Figure1.

Sixth Embodiment

In the Fifth Embodiment description "In a conventional 2-shaft style of gas turbine, the primary recycle compressor is shaft-connected to the high-pressure stage section of the power turbine assembly, and the low pressure section of the power turbine assembly provides the turbine power output power to driven equipment.", the presented invention provides alternative system and apparatus means by which an unconventional turbine power train (comprising individual separate compressor unit assembly, oxy-fuel combustion burner assembly, and hot gas expansion turbine assembly unit with mechanical shaft output) can be configured to produce mechanical

or electrical power within a cogeneration system as described later and shown in Figure 2.

The invention's primary recycle compressor can be a separately motor-driven or stream turbine-driven compressor of centrifugal or axial type therein comprising one or more stages of compression as required, or single rotating positive displacement type for the system applied operating conditions. The re-circulated and slightly superheated turbine exhaust gas stream is re-introduced into the primary recycle compressor and increased in pressure and temperature as described for the conventional gas turbine power system. This style of primary recycle gas compression drive train generally offers greatly improved capacity control and/or turn-down capabilities, but can be overall less efficient than the conventional gas turbine assembly's direct-driven axial compressor section.

As described in the Fourth and Fifth Embodiment, the oxy-fuel combustion burner assembly configuration and functional operation remains unchanged. Rather than the Fifth Embodiment described one or more oxy-fuel combustion burner assembly being positioned radially about the centerline axis of the power turbine assembly, the presented invention's alternative system and apparatus means can have a single oxy-fuel fired combustion burner assembly that is axially centerline-positioned and can be directed-connected to the hot gas expander power turbine as shown later in Figure 2. A single oxy-fuel combustion burner assembly can comprise multiple elements of existing manufactured oxy-fuel burner nozzle models rated from 0.6 to 14 MM Btu/Hr. as typically employed in the glass and steel making industries., or can comprise modifications to existing single industrial steam generation or process heater burner models that can be rated between 25 to 500 MM Btu/Hr.

Seventh Embodiment

From the Second Embodiment's cited ".....the said re-circulated turbine gas within the exhaust distribution manifold comprises the discharge exhaust gas from a second WHRSG or WHRPF exchanger upstream connected to a re-circulated exhaust gas manifold that conveys the combined turbine reduced temperature exhaust gases from both the WHRU exchanger and the first parallel-positioned WHRSG or WHRPF exchanger into which the gas turbine high temperature exhaust is first connected.", the total amount of waste heat that can usefully be transferred into the said heat exchanger's fluids is limited to or in proportion to) the amount of turbine output power that is developed by the invention's power cogeneration system turbine unit.

The presented invention provides alternative system and apparatus means by which a power turbine cogeneration system's production of steam or water (or heating of process fluids) is independent of the amount of turbine developed power within a cogeneration system. This presented invention, with its described alternative system and apparatus means, provides this cogeneration system with added operational flexibility while further increasing the thermal efficiency of the presented invention's cogeneration system and maintaining the same ultra-low exhaust emissions. Wherein a presented cogeneration system facility of a given power output rating could fully utilize a 100% or greater steam production or process fluid heating than would be associated with the cogeneration system and apparatus means shown in Fig. 1, the Fig. 2 presented alternative cogeneration system and apparatus means can include the presented supplementary oxy-fuel fired heating of recycled system exhaust gases to achieve the additional production of steam or process fluid heating while achieving the presented overall cogeneration system thermal efficiencies that can significantly exceed

115% as shown later in Table 5 for an example 100% increase in steam or process heating beyond the Fig. 1 system capabilities.

The presented invention's alternative system and apparatus means includes the added conduit means for withdrawal of re-circulated turbine exhaust gas from the Third Embodiment described exhaust gas distribution manifold for the conduit routed supply of the re-circulated turbine exhaust gas to the example preferred two parallel auxiliary primary recycle blowers that are separately capacity controlled to produce slightly re-pressurized first and second conduit stream flows of exhaust recycled gas that are connected to the alternative cogeneration system's auxiliary oxy-fuel combustion burner assembly unit. The oxy-fuel combustion burner assembly employs additional individual connected flow controlled streams of fuel and predominant oxygen to produce an identical composition of combustion exhaust gases as existing within the turbine exhaust gases, whereby the said added oxy-fuel combustion burner assembly's exhaust gases are conduit routed into the turbine exhaust conduit branch connecting to the WHRSG exchanger or WHRPG exchanger described above in the above cited Second Embodiment text.

In the case of the Fig. 1 configuration of the presented invention's cogeneration system and apparatus means, any increase in power generation (beyond the then existing cogeneration system's 'steady-state' production condition, but not exceeding the turbine's continuous rating) can be accomplished by terminating the controlled flow of vented excess turbine re-circulated exhaust flow to atmosphere and increasing fuel flow. Only upon reaching the required accumulated increased mass flow of preset high temperature exhaust gases within the closed system, is the presented invention's

power cogeneration system then returned to its normal 'steady-state' and 'partially-open system status' with controlled excess re-circulated exhaust gas vented to atmosphere.

Eighth Embodiment

From the First Embodiment cited "As shown in Table 1, between power turbine oxy-fuel combustion burner assembly pressures of 45 psia and 75 psia, the AES Cycle thermal efficiencies can range between 35.16% and 43.24%", the invention's improved high thermal efficient cogeneration system's presented example of a 60 psia oxy-fuel combustion burner assembly enables a low fuel gas supply pressure of less than 65 psi gage (5.5 Bar) to be employed.

Ninth Embodiment

From the preceding collective Embodiments' cited control of fluid stream flows, temperatures, pressures, generated power, and apparatus means includes valves, compressors, blowers, motors, etc., the presented invention's cogeneration system and apparatus means can be both performance and safety monitored and controlled by a manufacturer's PLC based control panel design that meets or exceeds the American Petroleum Institute (API) specifications for industrial gas turbines (API 616) or aero-derivative gas turbines specification (API RP 11PGT), or API 617 for centrifugal compressors (and applicable portions therein to be applied to hot gas expanders), or API 619 for rotary positive displacement compressors, or API 673 for special fans, or added safety monitoring as required within API 560 for fired heaters for general refinery service, or NFPA 85C for prevention of boiler and furnace explosions, and can be further control-integrated with a power plant distributive control system (DCS). The PLC based control panel design can further comply with other prevailing commercial, industrial or other governmental jurisdiction codes and standards. Other cogeneration

plant individual auxiliary support system modular component PLC control panel's operating output data signals can be control-integrated into the DCS together with the operating cogeneration power system's operating data signals comprising but not limited to:

(a) the cogeneration system's individual valve controlled gas stream's mass flows with temperatures and pressures for a given operating hydrocarbon fuel composition and horsepower or kilowatt output, and effective waste heat transfer duty;

(b) the cogeneration system's power turbine exhaust and waste heat recovery unit's fluid conditioning status and turbine exhaust excess oxygen content for a given operating hydrocarbon fuel composition;

(c) the cogeneration system's power turbine exhaust and primary recycle compressor discharge mass flow rates through their respective downstream waste heat recovery exchangers;

(d) the cogeneration facility's auxiliary rotating equipment's operating mass flow rates with temperatures and pressures combined with the positioning-state of any rotating equipment's integral capacity control apparatus;

(e) the cogeneration facility's rotating equipment and fired heater safety monitoring condition point locations as set forth by the prevailing industry or government specifications for each type of equipment, as well as those monitoring points whose operating condition state can impact on the cogeneration system's operational on-line availability and equipment life cycle costs.

Overall System and Apparatus Means

Within the presented partially-open turbine power cogeneration system and apparatus means described herein, the provided system employed oxy-fuel combustion generated working motive fluid means can provide a 95 to 100% reduction of nitrogen oxides (NO.sub.x) that occurs within current art Low-NO.sub.x gas turbines. The provided partially-open turbine power cogeneration system's temperature controlled oxy-fuel combustion temperature and the speed of combustion flame heat transfer also similarly suppresses the chemical reaction dissociation formation of the fugitive emission carbon monoxide (CO) from carbon dioxide (CO.sub.2). The means of suppressing the development of fugitive emissions results from the following collective working motive fluid molecular attributes and combustion events:

(a) The working motive fluid of this invention's power cogeneration system comprises a continuous superheated mixture of predominant carbon dioxide (CO.sub.2) and water vapor (H.sub.2 O) in identical Mol percent ratio proportions as these molecular components are produced from the combustion of a given fuel. For example, in the case of landfill gas, the working gas fluid contains a 1:1 ratio of 2 Mol carbon dioxide to 2 Mols water vapor in identical proportion to the products of stoichiometric oxygen combustion. The chemical reaction equation can be described as follows:

Working Motive Fluid + 1 Mol CH.sub.4 + 1 Mol CO.sub.2 + 2 Mols O.sub.2 =

2 Mol CO.sub.2 + 2 Mol H.sub.2 O + Heat + Working Motive Fluid.

In the example of methane gas fuels, the working fluid composition contains a ratio of 1 Mol CO.sub.2 to 2 Mols H.sub.2 O in identical proportion to the products of 105% stoichiometric oxygen combustion of methane fuel within the chemical reaction equation of:

Working Motive Fluid + 1 Mol CH.sub.4 + 2.1 Mols O.sub.2 = 1 Mol CO.sub.2 + 2 Mols H.sub.2 O + 0.1 Mol O.sub.2 + Heat + Working Motive Fluid;

(b) The invention's turbine power cogeneration system's working fluid provides the replacement mass flow means to the conventional open Brayton simple cycle's predominant diatomic non-emissive and non-radiant energy absorbing molecular nitrogen (N.sub.2) working fluid. The invention's replacement working motive fluid contains both predominant water vapor (with a binary lack of molecular symmetry) and a mass ratio of atomic weights of (16/2) = 8 and carbon dioxide with a mass ratio of atomic weights of (32/12) = 2.66, which denotes their susceptibility to high radiant energy emissivity and absorption. This compares to the nitrogen's mass ratio 14/14 = 1 which denotes nitrogen's minimal, if any, emissive and radiant energy absorbing abilities at any temperature;

(c) The invention's turbine power cycle system's controlled flow of working motive fluid provides the means for turbine combustion chemistry with a 900 % increase of binary molecular mass means susceptible to the fuel/oxidation exothermic chemical reactions being highly accelerated at the speed of light (186,000 miles a second). This enables the complete and rapid combustion of gaseous or liquid hydrocarbon fuels through the absorption and emissive radiant heat transfer of the fuels' combustion product's highly superheated binary carbon dioxide and binary water vapor molecules' heat energy that is emitted in their individual infrared spectral ranges. The radiant heat is transferred by radiant energy absorption into the combined greater mass identical proportions of identical composition gases contained within the working motive fluid blended within the pre-combustion gases and more predominantly contained in the

outer secondary zone surrounding the primary combustion flame zone. The extremely rapid rate at which the combustion product gases are lowered in temperature means there is inadequate time for the chemical disassociation reactions to occur, which produce carbon monoxide (CO), or other chemical reactions which produce nitrogen dioxide (NO.sub.2), in the presence of the highly elevated gas molecular temperatures above 2600° F to 2900° F;

(d) The First Embodiment recited oxy-fuel combustion burner assembly pre-mix sub-assemblies provides the means for homogeneous blending, wherein gaseous streams of working motive fluid and an oxygen-rich stream are further homogeneously blended for downstream mixing and ignition with the gaseous fuel stream. The gaseous fuel stream also comprises binary molecules of high susceptibility to high radiant energy absorption and emissivity, such as methane with a mass ratio of atomic weights of (16/4) = 4, ethane with a mass ratio of atomic weights of (24/4) = 6, propane with a mass ratio of atomic weights of (36/8) = 4.5, etc;

(e) The subsequent tertiary zone admission of a controlled-flow of Table 1 identified 1350° F superheated working motive fluid into the example 2400° F. burner assembly's primary oxy-fuel combined primary combustion flame zone and its outer secondary zone combustion gas stream, results in the rapid creation of the example desired equilibrium temperature of 1800° F. This rapid establishment of the preferred equilibrium temperature is due to the 186,000 miles per second rate of radiant heat transfer between the two streams of common molecular constituents with common means of high radiant energy absorption and emissivity in their respective individual infra-red spectrum ranges.

The presented cogeneration power system's oxy-fuel combustion system's generated working motive fluid of optimum selected operating pressures and temperatures can achieve 115% or greater cogeneration system thermal efficiencies. The means of achieving these 40% to 50% or greater thermal efficiencies than current art conventional cogeneration power facilities (thereby reducing CO.sub.2 "greenhouse emissions" by 40% to 50%), results from the following collective working fluid molecular attributes, system design, and apparatus features:

(a) The oxy-fuel combustion burner assembly's low operating pressures reduces the work (per pound of primary recycled gas) that is adsorbed by the turbine train's compressor section that re-pressurizes the recycled gas stream that becomes the downstream highly superheated working motive fluid that is expanded through the hot gas expansion turbine assembly;

(b) The presented power cogeneration system working motive fluid molecular gas composition replaces air content nitrogen that is the predominant mass flow component in the conventional gas turbine working motive fluid. The presented cogeneration system working fluid is unique in that each highly superheated temperature pound of fluid can adsorb or exchange 42% more heat per degree Fahrenheit change in gas temperature than does air or nitrogen.

(c) In the presented example operating conditions, approximately 92% of the high temperature turbine exhaust heat energy that is recovered from within the total exhaust flow passing through the WHRU exchanger and first WHRSG exchanger (or WHRPF exchanger) is transferred back into the low pressure working motive fluid that will re-enter the oxy-fuel combustion burner assembly to further absorb the heat of fuel combustion.

(d) Approximately 95% of the presented cogeneration system's re-circulated exhaust downstream of the waste heat exhaust exchangers (containing a large 'heat sink' quantity of energy) is recycled within the closed portion of the cogeneration system during steady-state operation. During an increased energy output demand on the presented power cogeneration system, 100% of the presented cogeneration system's re-circulated exhaust heat capacity downstream of the waste heat exhaust exchangers is recycled during its accompanying 'total-closed' cycle system operation.

With the presented partially-open turbine cycle power cogeneration system and apparatus means described herein, or presented alternative system and apparatus means, either a modified conventional gas turbine unit power train or an unconventional turbine power train can be employed. An alternative AES turbine assembly unit apparatus configuration can utilize separate existing low cost mechanical equipment components and burner assemblies which are predominantly not designed for, nor applied to, the manufacture of conventional gas turbines, nor the said components' incorporation into facility designs of current technology gas turbine power cogeneration systems.

Within the presented partially-open turbine cycle power cogeneration system and apparatus means described herein, the presented invention provides alternative system and apparatus means by which a turbine cogeneration system's production of steam or water (or heating of process fluids) is independent of the actual percentage of rated electric power load that is being produced from an operating turbine powered cogeneration system. The presented alternative system and apparatus means is not limited in its ability to have expanded steam or hot water or process heating capacity means beyond that which is possible solely from turbine waste heat utilization.

Within the presented partially-open turbine cycle power cogeneration system and apparatus means described herein, the systems and apparatus means are provided wherein all fluid streams entering the oxy-fuel fuel combustion burner assembly are controlled to maintain preset maximum combined primary combustion flame zone and outer secondary zone temperatures in which a non-distribution quality of gaseous hydrocarbon fuel (containing toxic and/or difficult to combust hydrocarbon molecular gases) can be rapidly carried through oxy-fuel combustion to a useful heat conversion and/or completed incineration without altering the system's high thermal efficiencies or ultra-low emission levels.

BRIEF DESCRIPTION OF THE DRAWINGS

Fig.1 is a schematic flow diagram of the invention's improved cogeneration system and apparatus that includes the presented AES partially-open power cycle with a modified configuration of a conventional gas turbine and simplified waste heat transfer for steam generation or process fluid heating.

Fig.2 is a schematic flow diagram of the invention's improved cogeneration system that includes the presented AES partially-open power cycle system and apparatus of Fig.1 and additional alternative apparatus means including an alternate separate motor or steam turbine driven recycle compressor, oxy-fuel combustion burner assembly series-connected to a hot gas expander turbine, and alternative supplementary oxy-fuel combustion burner assembly that increases system steam or hot water production or heating of process fluids.

DETAILED DESCRIPTION OF THE PREFERRED EMBODIMENT

Referring now more particularly to Fig. 1, a modified conventional gas turbine's exhaust recycle gas compressor section **1** comprises 2 or more recycled exhaust gas compression stages, positioned in series, with a final stage of radially directed discharge flow of compressed recycle exhaust gas. In the case of a two-shaft turbine, the power to drive the recycle compressor section **1** is transmitted by shaft **2**, on which one or more high-pressure power extraction turbine stages are mounted within the combustion hot gas expansion power turbine assembly **3**. The second shaft, designed for mechanical equipment or generator drive applications, has one or more low-pressure hot gas expansion stages mounted on power output shaft **4**, with coupling means for power transmission to rotate the driven equipment.

The invention's cycle adaptation to conventional gas turbine driven mechanical equipment may or may not require the addition of a gearbox or variable speed coupling **5** to adapt the speed of the hot gas expansion power turbine **3** to the speed required by a generator or other driven equipment (not shown). The rotating driven equipment may have its required power transmitted through a shaft and coupling means **6**. The shaft and coupling means **6** can transmit power to a generator **7**, wherein electric power is produced and transmitted through conduit means **8** to a control room module **9**. Control room module **9** therein contains the modular turbine unit's PLC control panel and switchgear and motor control center, whereby electric power production is controlled and distributed to the power facility's electrical grid and/or connected electric utility electrical grid. The shaft and coupling means **6** may alternately transmit power to other rotating pumps or compressors in lieu of generator **7**.

Within the presented invention's partially-open power cogeneration system, the slightly superheated turbine exhaust re-circulated gas flows from the turbine exhaust gas distribution manifold **10** (having end-connection **62** that is blind-flanged closed in this Figure 1) through said manifold side-branch connected turbine exhaust recycle gas conduit means **11** that is end-connected to the inlet of the turbine exhaust gas primary recycle compressor section **1**. The higher-pressure and higher-temperature compressed recycle turbine exhaust gas (hereafter referred to as "primary re-pressurized recycle gas") is routed through conduit manifold **12** containing two parallel conduit end-branches **13** and **14** respectively, each said branch containing a gas mass flow sensor means and a flow control (or flow proportioning) damper valve **15**.

The twin conduit end-branches **13** and **14** respectively convey first and second primary re-pressurized recycle gas streams with respective end connections to parallel inlet headers **16** and **17** located on the primary section **18** of the power turbine exhaust gas WHRU exchanger. The said first and second streams of primary re-pressurized recycle gas is discharged from primary section **18** of the power turbine exhaust gas waste heat recovery unit (WHRU) exchanger through outlet headers **20** and **19** respectively at highly increased superheated temperatures (with the highly superheated recycle gas hereinafter referred to as a "working motive fluid") with flows through conduits **21** and **22** respectively.

The primary re-pressurized recycle gas is additionally routed at low gas flow levels from conduit manifold means **12** through a side-branch connected conduit means **23** containing motor driven air-cooler **24** and flow control valve **25** for subsequent downstream conduit end-connection to one or more partial premix sub-assemblies **27**

contained within one or more oxy-fuel combustion burner assembly **26** that can be positioned radially about the centerline axis of the power turbine assembly.

Conduit **22** conveys the second controlled stream of working motive fluid to the internal primary combustion zone **28** contained within each oxy-fuel combustion burner assembly **26.** Conduit **21** conveys the first controlled stream of working motive fluid to the internal tertiary blending zone **29** contained within each oxy-fuel combustion burner assembly **26** that can be positioned radially about the centerline axis of the turbine assembly. The combined streams of working motive fluid composition gases exiting tertiary blending zone **29** can be routed through conduit flow means **30** having end connection to the inlet of power turbine assembly **3.**

Alternately the conduit **21** can convey the first controlled stream of working motive fluid to a common single tertiary blending zone that receives primary combustion zone working fluid composition gases from two or more oxy-fuel fired combustion burner assembly **26** that is positioned immediately upstream of the described alternate single common (not shown) tertiary blending zone. The combined streams of working motive fluid composition gases exiting the common tertiary blending zone (not shown) are routed through conduit **30** having end connection to the inlet of power turbine assembly **3**.

A pressurized stream of presented example methane fuel gas (or alternate acceptable liquid hydrocarbon fuel) is supplied from source **31** into conduit **32** containing sensor-transmitter means for temperature, pressure, mass flow, and fuel flow control valve means **33**, with said conduit having end–connectivity to either one or more preferred downstream partial pre-mix subassembly **27** contained within oxy-fuel combustion burner assembly **26.**

A controlled pressurized stream of predominant oxygen is supplied from a facility remote source **34** into conduit **35** containing sensor-transmitter means for temperature, pressure, mass flow, and flow control valve means **36**, with said conduit having end-connectivity to either one or more preferred partial pre-mix subassembly **27** contained within oxy-fuel combustion burner assembly **26.**

Within the partial pre-mix subassembly **27**, the said identified conduits **23**, **32**, and **35** respectively supplied controlled stream flows of primary re-pressurized recycle gas, fuel, and predominant oxygen are therein partially blended therein for following downstream ignition and controlled temperature combustion within the temperature sensor-transmitter monitored primary combustion zone **28** therein having further admitted second controlled stream of working motive fluid composition gases supplied by conduit **22.**

Within oxy-fuel fired combustion burner assembly **26**, the combined mass flows of products of fuel combustion and streams of working motive fluid composition gases flows from the primary combustion zone **28** at a controlled highly superheated presented example equilibrium temperature of 2400F into the downstream positioned tertiary blending zone **29** wherein these said gases are blended with the controlled mass flow of fore-described conduit **21** supplied first stream of working motive fluid composition gases.

The combined working motive fluid composition gases' mass flows entering the tertiary blending zone **29** within oxy-fuel fired combustion burner assembly **26** therein produces a resultant selected equilibrium temperature and mass flow rate of superheated gases through conduit **30** into the hot gas expander power turbine subassembly **3**. Work is developed within the hot gas expander power turbine

subassembly **3,** and the heat energy or enthalpy (Btu/lb) contained within the mass flow of expanded exhausted gases is decreased and discharged into conduit **37**. Conduit **37** routes the hot gas expander power turbine subassembly exhaust gases through conduit end-branches **38** and **41** that are respectively connected to WHRU exchanger **18** and waste heat recovery steam generator (WHRSG) or waste heat recovery process fluid heater (WHRPF) exchanger **42.** The proportional division of the total mass flow of the hot gas expander power turbine subassembly **3** exhaust gas contained within conduit **37,** between conduit end-branches **38** and **41,** is controlled or flow-proportioned respectively by damper valves **40** and **44** contained within the WHRU exchanger **18** and WHRSG or WHRPF exchanger **42** respective outlet exhaust branch conduits **39** and **43.** The predominant portion of conduit **37**'s total mass flow of exhaust gases is divided and directed through WHRU exchanger **18** to satisfy the exhaust heat transfer requirements to the primary re-pressurized recycled gas flowing through exchanger **18**.

In the case of waste heat transfer to a facilities supplied hot water/steam or process fluid circuit, a pressurized stream of a cogeneration facility's steam condensate feed water (or process fluid) can be supplied from source **46** into conduit **47** that can contain sensor-transmitter means for both temperature and mass flow, and having end-connectivity to the inlet header **48** of a second (WHRSG) or WHRPF exchanger **49**. In the case of stream generation, the supplied stream of steam condensate can be changed from a liquid phase to a liquid/vapor 2-phase state or slight superheated steam vapor state within exchanger **49**, and exits from exchanger **49** through discharge header **50** into conduit **51** having end-connectivity to the inlet header **52** of WHRSG exchanger **42**. Within WHRSG exchanger **42**, the steam circuit stream can be highly superheated as desired to a cogeneration system produced steam superheat

temperature ranging from less than 900°F to over 1200°F for discharge from outlet header **53** into conduit **54** that can deliver the superheated steam to a facility delivery connection point **55.** For the alternative addition of increased cogeneration system mass flow steam generation (as described later in Figure 2), expander turbine exhaust gas conduit **37**'s end-branch conduit **41** can be supplied with a connected side-branch conduct **56** whose end connection **57** that is closed with a blind-flange in Figure 1.

The cogeneration system's reduced temperature exhaust gases exits from the WHRU exchanger **18** and the parallel-positioned WHRSG exchanger or WHRPF exchanger **42** through their respective exhaust gas discharge branch conduits **39** and **43**, each branch conduit respectively therein containing controlled-flow damper valves **40** and **44**. The reduced temperature system exhaust gas flows from branch conduits **40** and **44** are combined within re-circulated exhaust gas manifold **45** having end-connectivity to a downstream-positioned second WHRSG exchanger or WHRPF exchanger **49.** The system's re-circulated exhaust gases are reduced in temperature within the second WHRSG exchanger or WHRPF exchanger **49** to a temperature that is slightly above the dew point temperature of the re-circulated exhaust gas as it is discharged from the heat exchanger **49** into the exhaust gas distribution manifold **10**.

Within the presented invention's partially-open cogeneration power system, the slightly superheated turbine re-circulated exhaust gas mass flow within exhaust gas distribution manifold **10** remains at a constant flow rate for steady-state cogeneration thermal energy conversion. The excess slightly superheated turbine re-circulated exhaust gas mass flow within manifold **10** that is not required for steady-state turbine power production is flow-directed from manifold **10** through side-branch conduit **58** therein containing back pressure control valve **59** and flow control valve **60** and having

downstream connectivity to atmosphere at vent point **61**. The terminal end of exhaust gas distribution manifold **11** is provided with a closed blind flange connection **62** in Fig.1.

Fig.2 is a schematic flow diagram of the invention's improved cogeneration system that shows the same presented partially-open power turbine cycle system as shown in Fig. 1 with added specifically herein described alternative apparatus means that can include both an alternate separate motor or steam turbine driven recycle compressor and industrial-type oxy-fuel combustion burner assembly that is series-connected to a separate hot gas expander turbine with output power connection means. Fig.2 further shows and describes the alternate system addition of a separate oxy-fuel combustion burner assembly that performs the function of a supplementary hot exhaust gas generator to increase the cogeneration system's production of steam or the heating of process fluids.

Referring now more particularly to Fig. 2, the presented invention's improved cogeneration system therein incorporates the AES partially-open power cycle system and alternative apparatus means that can include an alternative separately driven primary recycle compressor comprising two or more power system recycle gas compression stages, with a final stage radially-directed discharge flow of primary re-pressurized recycle gas. Primary recycle compressor **63** can alternately be directly driven by an electric motor or steam turbine type driver **64**, or indirectly-driven through either gearbox or variable speed coupling means **65.**

The system's hot gas expansion power turbine assembly **67** can comprise one or more power extraction turbine stages and assembly output shaft that can be directly connected to electrical generator **7** wherein electric power is produced and transmitted

through conduit means **8** to a control room module **9**. Control room module **9** therein contains the power cogeneration system's PLC control panel, switchgear and motor control center, whereby electric power production can be controlled and distributed to the operating facility's electrical grid and/or to the utility electrical grid. Alternately (not shown), a gearbox or variable speed coupling can be positioned between the power turbine assembly output shaft and alternative driven rotating pumps or compressors (not shown) in lieu of generator **7**.

Within the presented invention's partially-open power cogeneration system of Fig.1, the slightly superheated turbine exhaust recycle gas can flow from the turbine exhaust gas distribution manifold **10** with exiting flows through open end-connection **62** that series-connects to manifold extension conduit **68** as further described later. Manifold **10** side-branch connected turbine exhaust recycle gas conduit means **11** is end-connected to the inlet of the turbine exhaust gas primary recycle compressor **63**. The higher-pressure and higher-temperature re-pressurized recycle turbine exhaust gas (hereafter referred to as "primary re-pressurized recycle gas") and related identical stream flows are thereafter the same as described as in Fig.1 for its routing through WHRU **18** and continuing to oxy-fuel fired combustion burner assembly **26.** The hot gases generated within oxy-fuel fired combustion burner assembly **26** are routed through direct-connected gas transition assembly **66** with end connectivity to the inlet of power turbine assembly **67**.

Conduit **37** routes the hot gas expander turbine assembly **67** exhaust gases through conduit end-branches **38** and **41** that are respectively connected to WHRU exchanger **18** and waste heat recovery steam generator (WHRSG) or process fluid heat exchanger **42** and thereafter described associated conduit streams are as described for Fig.1. For

the alternative addition of increased cogeneration system mass flow steam generation, fore-described conduit **68** can route a flow of slightly superheated turbine exhaust recycle gas through preferred parallel end-branch conduits **69** and **70** respectively containing flow control provided isolation/damper valves **71** and **72** and having end connectivity with one or more parallel-positioned **73** and **74** speed-controlled motor-driven exhaust recycle exhaust gas blowers. Exhaust recycle gas blower **73** provides a required mass flow of exhaust recycle gas at a slightly increased pressure into its discharge conduit **75** having end-connectivity with the tertiary blending zone **82** contained within the downstream-positioned oxy-fuel fired burner assembly **79.** Exhaust recycle gas blower **74** provides a required mass flow of exhaust recycle gas at a slightly increased pressure into its discharge conduit **76** having end-connectivity with the partial pre-mix subassembly **80** contained within the downstream-positioned oxy-fuel fired combustion burner assembly **79.**

A controlled stream of low pressure predominant oxygen is supplied from facility remote source **77** into conduit **84** containing sensor-transmitter means for temperature, pressure, mass flow, and oxygen flow control valve means **85**, with said conduit **84** having end-connectivity to either one or more preferred partial pre-mix subassembly **80** contained within oxy-fuel fired combustion burner assembly **79.**

A low pressure stream of presented example methane fuel gas (or alternate acceptable liquid hydrocarbon fuel) is supplied from source **78** into conduit **86** containing sensor-transmitter means for temperature, pressure, mass flow, and fuel pressure/flow control valve means **87**, with said conduit **86** having end–connectivity to either one or more downstream-positioned preferred partial-premix subassembly **80** contained within oxy-fuel fired combustion burner assembly **79.**

Within the partial pre-mix subassembly **80**, the said identified conduits **76**, **86**, and **84** respectively supplied stream flows of turbine exhaust recycle gas, fuel, and predominant oxygen are therein blended for following downstream ignition and controlled temperature combustion within the temperature sensor-transmitter monitored primary combustion zone **81** contained within oxy-fuel fired combustion burner assembly **79.**

Within oxy-fuel fired combustion burner assembly **79**, the predominant mass flow of combined products of fuel combustion and turbine exhaust recycled gas flows from the primary combustion zone **81** (at a controlled high superheated presented example equilibrium temperature of 2400F) into the downstream tertiary blending zone **82** wherein these said composition gases can be blended with the controlled mass flow of fore-described conduit **75** supplied blower discharge stream of slightly re-pressurized and low superheated power turbine exhaust recycle gases of identical gas composition.

The oxy-fuel fired combustion burner assembly **79** provides a supplementary flow of slightly re-pressurized and highly superheated turbine recycle exhaust gas mass flow at controlled temperatures into conduit **83** having end connectivity to conduit **56**'s flanged connection **57**. The supplementary flow of slightly re-pressurized and highly superheated turbine recycle exhaust gas mass flow is routed through conduit **56** into branch conduit **41** having connectivity to WHRSG exchanger or process fluid exchanger **42**, thereby enabling a increased mass flow of steam or hot water or process fluids (in conduits **47**, **51**, and **54** at given desired temperature operating conditions) to be additionally generated with high system thermal efficiency within the WHRSG or process fluid exchangers **49** and **42** from the invention's increased cogeneration system's increased mass flows of superheated recycled exhaust mass flows.

Within the presented invention's partially-open power cogeneration system, the slightly superheated turbine recycle exhaust gas mass flow within conduit **11** remains at a constant flow rate for steady-state turbine power shaft horsepower output production. The excess slightly superheated turbine recycle exhaust gas mass flow within manifold **10** that is not required for steady-state turbine power production, nor is required to maintain an existing steady-state recycle exhaust gas mass flow rate within conduit **68** for the fired oxy-fuel combustion heater **79**, is flow-directed from manifold 10 through side-branch conduit **58** containing back pressure control valve **59** and flow control/isolation valve **60** with downstream connectivity to atmosphere occurring at vent point **61**.

The numbers in Table 2 below are representative of: one example set of fluid stream conditions in which the AES turbine power cycle portion within the presented cogeneration system can operate (the conduit streams are those identified by the numbers in Fig. 1). The following assumptions were made: both the recycle compressor efficiency and hot gas expansion turbine efficiency is 84%; the oxy-fuel combustion burner assembly operating pressure is 60 psia; and the methane fuel gas flow rate is 1 Mol/minute.

TABLE 2

Conduit Stream Number	Stream Fluid	Temperature Degree F.	Pressure PSIA	Mass Flow lbs./Min.
11	Recycle Exhaust	197	15	1879
12	Compressed Recycle	500	64	1879
22	WMF – Primary Zone	1350	63	686

21	WMF – Tertiary Zone	1350	63	1153
23	Cooled Compressed Recycle	280	63.5	40
32	Methane Fuel	70	85	16
35	Predominant O.sub.2	110	65	64
30	Combustion Gas	1800	60	1959
37	Turbine Exhaust	1391	15.8	1959
45	WHRU&WHRSG Exhaust	530	15.4	1959
58	Cogen System Vent Gas	197	15.1	81

(WMF) = Working Motive Fluid

With the same example stream conditions and assumptions made for Table 2, supra, Table 3 provides the thermodynamic values from which the tabulated compressor horsepowers and turbine power outputs are derived.

TABLE 3

Conduit Stream ** Number	Rotating Equipment Name	Stream Fluid	Temperature Degrees F	Mass Flow lbs./Min.	Delta Enthalpy Btu/Lb.	Horse-Power (HP)
11 to 12	Exhaust Recycle Compressor	Inlet	197	1879	98.9	4377
		Discharge	500			
30 to 37	Hot Gas Expander Turbine	Inlet	1800	1959	169.7	7837
		Discharge	1391			
				Net Shaft Horsepower Output		3460 SHP *

(*) Note: (20,693,400 LHV Btu/Hr-Mol CH4) ÷ 3460 SHP = 5980 Btu/Hp-hr. fuel rate.
(*) Note: Fuel Rate: (2545 Bt/Hp-hr. ÷ 5980 Btu/Hp-Hr. = 42.55% turbine thermal efficiency:
(**) Note: Only the conduit stream numbers reference to both Figure 1 and Figure 2 drawings.

With the same conditions and assumptions made for Table 2, supra, Table 4 contains six conduit streams (as noted) that appear in both Fig. 1 and Fig. 2, with the thermal heat transfers and mass flow rates pertaining only to the Fig. 1 presented system.

TABLE 4

Conduit Stream Number	Heat Exchanger Name	Stream Fluid	Temperature Change Degrees F	Mass Flow lbs./Min.	Delta Enthalpy Btu/Lb.	Recovered Heat Rate Btu/Min.
37 to 45	18 + 42	Total Exhaust	1391F to 530F	1959	326	638,634
38 to 39	WHRU 18	Exhaust Gas	1391F to 530F	1805.15	326	588,480
13/14 - 21/22	WHRU 18	'WMF' Gas	500F to 1350F	1839	320	588,480
41 to 43	WHRSG 42	Exhaust	1391F to 530F	153.85	326	50,154 *
45 to 10	WHRSG 49	Exhaust	530F to 197F	1959	110	215,490 *

*Total Available Heat for Process Gas or Steam Circuit= (215,490 + 50,154) = 265,644 Btu/Min.
*Total Available Heat for Process Gas or Steam Circuit= (215,490 + 50,154) = 15,938,640 Btu/Hr.
Total 910 Btu/SCF LHV of 1 Mol/Min. Methane Fuel Gas = 344,890 Btu/Min. = 20,693,400 Btu/Hr.
Recovered Heat from the Supplied Fuel Gas Energy:
= (15,938,640 Btu/Hr ÷ 20,693,400 LHV Btu/Hr-Mol Methane Gas) = 77.02%.
Total Cogeneration System Thermal Efficiency:
= 42.5% Simple Cycle Turbine + 77.02% Recovered Heat = 119.5%.

With the same conditions and assumptions made for Table 2 and 4 supra, Table 5 provides the thermal heat transfers and mass flow rates as contained the Alternative Cogeneration System of Fig.2 with added supplementary heat blended into the turbine exhaust stream to increase the cogeneration system's effective transfer of heat to steam or process heated fluids by the example amount of 100%.

TABLE 5

Conduit Stream Number	Heat Exchanger Name	Stream Gas	Temperature Change Degrees F	Mass Flow lbs./Min.	Delta Enthalpy Btu/Lb.	Recovered Heat Rate Btu/Min.
38 to 39	WHRU 18	Turbine Exh.	1391F to 530F	1805	326	588,480
13/14 - 21/22	WHRU 18	'WMF' Gas	500F to 1350F	1839	320	588,480
41/83 - 43	WHRSG 42	Exhaust	1391F to 530F	763	326	248,738 *
45 to 10	WHRSG 49	Exhaust	530F to 197F	2568	110	282,480 *
10 to11		Recycle		1879		
10 to 68		Recycle	197F	556		
10 to 61		Exhaust Vent		138		
35 + 84		95% Oxygen Mixture	120F	112		
32 +86		Methane Fuel	70F	26		

*Total Available Effective Energy Conversion to Heat for Process Gas or Water/Steam Circuit:

= (248,738+ 282,480) = 531,218 Btu/Min. = 31,873,080 Btu/Hr.

Turbine Power Apparatus Effective Energy Conversion Rate = (2545)x(3460 SHP) = 8,805,700 Btu/Hr.

Total Effective Energy Conversion Rate = 40,678,780 Btu/Hr.

Total System Fuel Energy Consumption:
(20,693,400 LHV Btu/Hr. for Turbine Apparatus+ 12,993,602 LHV Btu/Hr for Supplementary AES Burner System) = 33,687,002 LHV Btu/Hr.
Overall System Thermal Efficiency: (40,678,780 Btu/Hr.) ÷ (33,687,002) = 120.75%

It should be understood that the forgoing description is only illustrative of the invention. Various system and apparatus alternatives, fuels, and modifications to operating conditions can be devised by those skilled in the art without departing from the invention. Accordingly, the present invention is intended to embrace all such alternatives, modifications and variances which fall with the scope of the following appended claims.